

2000

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Seve, F.; Berlioz, A.; Dufour, R.; Charreyron, M.; Peyaud, F.; and Audouy, L., "Balancing of a Variable Speed Rotary Compressor: Experimental and Numerical Investigations" (2000). *International Compressor Engineering Conference*. Paper 1470.
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BALANCING OF A VARIABLE SPEED ROTARY COMPRESSOR: EXPERIMENTAL AND NUMERICAL INVESTIGATIONS

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Abstract

The article focuses on the balancing of a hermetic rotary compressor capable of variable speed of rotation. It follows studies in which an FE model was established. The FE model, based on the rotor-dynamics theory, is used in connection with the influence coefficient method. Numerical and experimental unbalance responses obtained with different balancing criteria are compared. It is shown that efficient balancing of the compressor must take into account the flexibility of the rotor-crankshaft sub-assembly and the motion of the hermetic housing. The proposed balancing satisfies both the vibration level minimization criterion and its industrial implementation.

INTRODUCTION

Nowadays, refrigerant rotary compressors running continuously at variable speed of rotation are of particular interest, since their BTU production can be adapted to demand and their lifetime prolonged due to the reduction of major transient phenomena caused by start-stop cycles, which are source of mechanical failures.

The rotary compressor studied (cf. Figure 1) consists of rotating and non-rotating mechanical parts. The rotor-crankshaft sub-assembly, equipped with the rolling piston, constitutes the rotating part while the electric stator, the crankcase, the hermetic housing are the main non-rotating part. The rotor-crankshaft sub-assembly is mounted in the two-bearing crankcase that contains the pump. The crankcase is welded to the housing by three brackets and the electric stator is press-fitted into the housing.

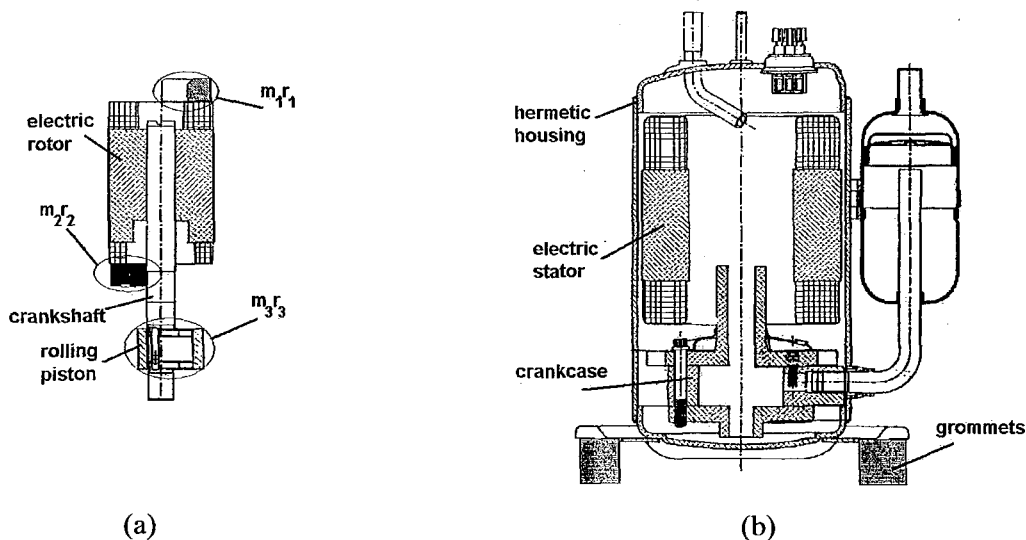


Figure 1. Rotating (a) and non-rotating (b) parts of the rotary compressor

Let m_i and r_i be respectively the mass and radius of gyration of the eccentric mass i . The current counterweight masses located at the top (m_1r_1) and the bottom (m_2r_2) of the rotor are designed to reduce the vibration level generated by the eccentric masses (m_3r_3) of the pump (crankpin, rolling piston, vane). However, mechanical problems occur at the highest speeds of rotation: rotor-stator and crankshaft-bearing rubs, bracket weld spot failures. Previous studies [1-4] establishing a finite element model based on the rotor dynamics theory showed that this is due to the overhung electric rotor, crankshaft flexibility and inefficient current balancing.

The aim of this study is to establish a procedure that takes into account crankshaft flexibility and housing motion to obtain efficient balancing throughout the operating speed range. After a basic definition of balancing methods, the influence coefficient method (ICM) was chosen and applied to the rotary compressor. Unbalance responses are predicted with both the current and new balancings. The associated experimental investigation uses a prototype compressor equipped with proximity probes to measure the lateral deflection of the housing.

1- BALANCING METHOD

Balancing rotors consists in minimizing the effect of unbalance masses on vibration levels and consequently on the bearing loads of the rotor. Considering the basic conclusions of the previous studies (see [4]), particular attention is paid to flexible rotor balancing methods [5] of which the main ones are the modal balancing method and the ICM. The former uses the modal parameters and the modal decomposition of the initial unbalance distribution, the rotor being balanced on its critical speeds, step by step, mode after mode [6]. This method is mostly used for balancing high speed machinery with considerable critical speed. Another way of achieving this is by using the ICM. This consists in evaluating the influence of trial masses on the orbits of the planes to be balanced, named target planes here. The solution of the associated inverse problem gives the ideal counterweight masses that minimize the vibration levels of the target planes [7, 8]. The balancing speeds of the ICM can be either critical speeds or any other speeds. This method is chosen for balancing the rotary compressor under study in the 1,200-7,200 rpm speed range which, moreover, does not contain any resonance phenomena.

Presentation of the influence coefficient method (ICM)

The aim of this method is to obtain the counterweight masses that minimize the amplitude of the target planes' orbits at different balancing speeds (cf. Figure 2). The ICM can be performed experimentally by measuring the orbits with sensors and fixing trial masses on the balancing planes. However, in rotary compressors, the eccentric mass of the mechanical parts of pumps are quite well known and its role is predominant in comparison to any initial unbalance distribution due to geometric or material uniformity faults. Therefore this investigation uses ICM performed numerically. All that is required is an accurate model of the machine to carry out the necessary successive simulations.

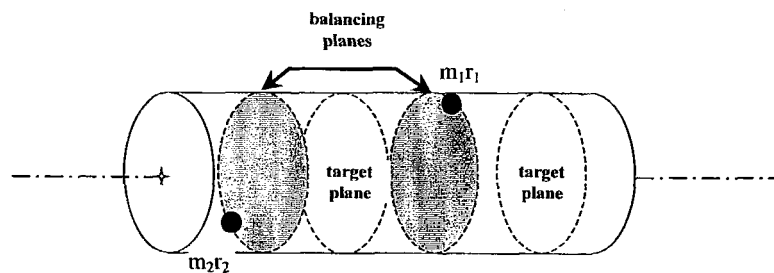


Figure 2. Balancing and target plane definition

Step one consists in choosing q balancing planes where the counterweight masses should be located, p target planes where the vibration level has to be minimized and s balancing speeds. If $q = p \times s$, the residual vibration levels are nil. If $q < p \times s$, the amplitudes cannot be nil but reduced. The efficiency of the balancing depends on the position and number of target planes and also on the number of balancing speeds, which must be chosen with care. The following steps consist in predicting the mass unbalance response of the rotor for several additional trial masses to determine the rectangular influence coefficient matrix A ($p \times s, q$). Then, each coefficient $a_{\ell m}^k$ of A , where $k = 1, \dots, s$, $\ell = 1, \dots, p$ and $m = 1, \dots, q$, which represents the influence of trial mass m on the vibration level of target plane ℓ at the balancing speed of rotation Ω_k , is calculated according to the formula:

$$a_{\ell m}^k = \frac{\bar{\eta}_m^{i+t}(y_\ell, \Omega_k) - \bar{\eta}^i(y_\ell, \Omega_k)}{\bar{U}_m^t} \quad (1)$$

with $\bar{\eta}^i(y_\ell, \Omega_k)$, the calculated complex amplitude of the initial vibration at speed Ω_k in the target plane ℓ having the y_ℓ abscissa, $\bar{\eta}_m^{i+t}(y_\ell, \Omega_k)$, the calculated complex amplitude of the vibration at speed Ω_k in the target plane ℓ with a trial counterweight mass $\bar{U}_m^t = m_m r_m e^{j\theta_m}$ added to the balancing plane m . Parameters m_m , r_m , θ_m , are respectively the mass, the radius of gyration and the angular position of the trial mass m . The last step is devoted to the calculation of the q counterweight masses \bar{U}_m^c . In the case where $q < p \times s$, matrix A is not square and no inversion is possible. Therefore a least square technique [7] is retained and yields the following system:

$$\{\bar{U}_m^c\} = -\{A^*\}^T [A]^{-1} [A^*]^T \{\bar{\eta}^i(y_\ell, \Omega_k)\} \quad (2)$$

where A^* is the conjugate of A .

Balancing of a refrigerant compressor with the ICM

The current balancing has been designed for 50/60 Hz operating speeds where the crankshaft can be assumed to be a rigid structure. Increasing the speed of rotation leads to over-pronounced deflection of the rotor-crankshaft sub-assembly and excessive vibration levels of the housing due to balancing made inadequate. Consequently, the balancing criteria are based on the minimization of the vibration level of the rotating and non-rotating parts.

Industrial requirements make it necessary to keep the two existing balancing planes at the top and the bottom of the electric rotor. A maximum of four target planes is used. They can be located either on rotating, or non-rotating parts. Thus the vibration levels of non-rotating part can be utilized in the present investigation. The more target planes there are, the higher the residual vibrations in each plane become. Two balancing speeds have been chosen, for example, 3,600 and 7,200 rpm.

Table 1 lists the four balancings tested. Balancing A is the current one. Balancing B uses three target planes of the rotor-crankshaft sub-assembly, balancing C only two target planes of the housing, and balancing D four target planes of the rotating and non-rotating parts. The reference phase (0°) is attributed to the eccentric mass of the pump.

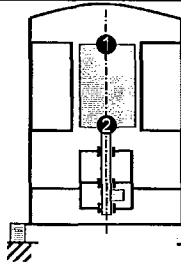
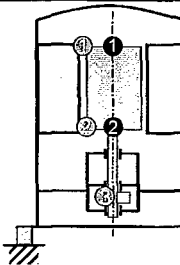
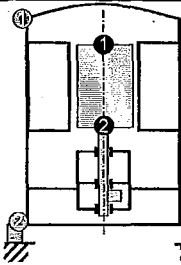
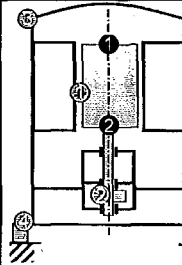
Balancing	A	B	C	D
❶ : balancing planes ❷ : target planes				
Balancing method	Current	ICM	ICM	ICM
Experimental validation	Yes	Yes	See balancing A	Yes
Counterweight masses	❶ : 435 g.mm at 0° ❷ : 1233 g.mm at 180°	❶ : 15 g.mm at 20° ❷ : 43 g.mm at 100°	❶ : 450 g.mm at -4° ❷ : 1319 g.mm at 175° Equivalent to balancing A	Counterweight retained

Table 1: Description of the balancing

2- NUMERICAL AND EXPERIMENTAL INVESTIGATIONS

The unbalance response obtained with balancings B, C and D are successively compared to the current one (balancing A). Numerical and experimental lateral displacements are presented using a dimensionless form. The displacement of the rotating part is referenced to the air-gap or bearing clearances. The experimental validation is highlighted on the orbits of the housing.

A and B balancings

Numerical investigation

Figure 3 presents the predicted lateral displacements versus speed of rotation for the A and B balancings, at the top of the rotor, at the main bearing (sections # 1 and # 2), and at the outboard bearing. It is shown that balancing B performed with the ICM substantially improves the response of the rotating part. Rotor to stator and crankshaft to bearing rubs vanish in the 1,200-7,200 rpm speed range. However, Figure 4 highlights an over-pronounced lateral displacement of the housing, in particular in the resonance phenomena caused by one of the rigid body modes of the compressor on its grommets. It should be noticed that the balancing B does not take into account the vibration level of the housing. Moreover, in balancing A, the displacements of the rotating and non-rotating parts are out-of-phase so that they cancel themselves, contrary to balancing B.

Experimental validation

Figure 5 compares the measured and predicted lateral displacement along the vertical axis of the housing at 4,800 rpm and shows a satisfactory experimental validation of the model. This has been checked for balancing B throughout the 1,200-7,200 rpm speed range; however, for the balancing A the maximum experimental speed reached was 5,400 rpm due to contact problems. The experimental investigation confirms the over-pronounced displacement of the housing: housing displacements must be considered in the balancing procedure.

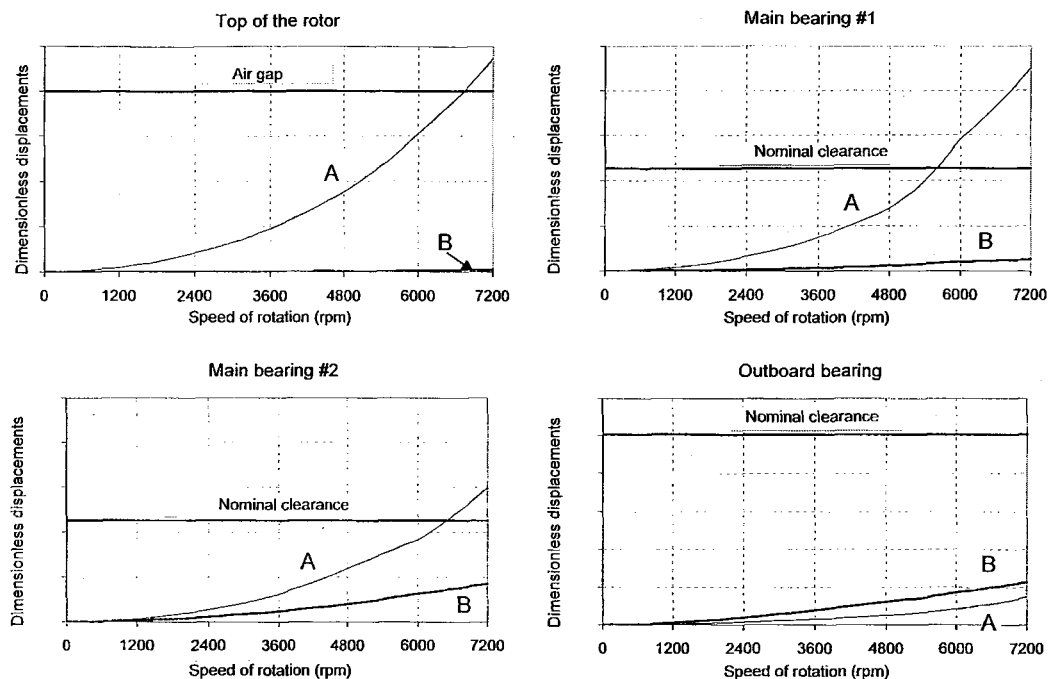


Figure 3: Displacements of the rotor-crankshaft sub-assembly – A and B balancings

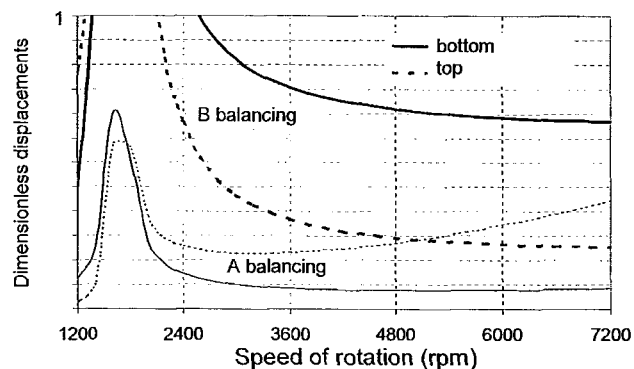


Figure 4: Displacements of the housing – A and B balancings

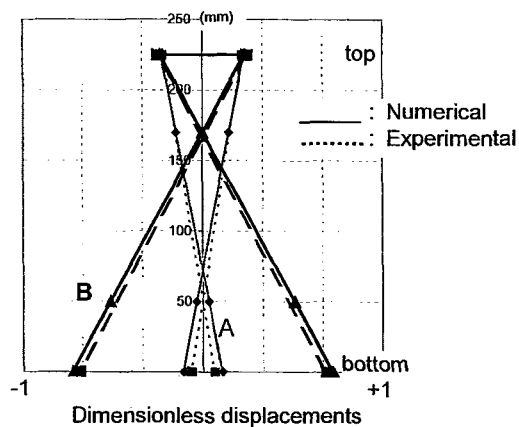


Figure 5: Dimensionless lateral displacement along the housing – A and B balancings

A and C balancings

Balancing C is performed with only two target planes located at the top and the bottom of the housing. The predicted counterweight masses are approximately the same as those of the current balancing.

A and D balancings

Numerical investigation

Figure 6 presents the predicted lateral displacements versus speed of rotation for the A and D balancings, at the top of the rotor, at the main bearing (sections # 1 and # 2), and at the outboard bearing. The lateral displacements of the rotor-crankshaft sub-assembly, although greater than those obtained with balancing B, are not over-pronounced. Figure 7 shows displacements at the top and the bottom of the housing and shows in particular the resonance phenomena due to one of the rigid body modes of the compressor on its grommets. The improvement of the vibration level of the housing is available in the 1,200-7,200 rpm speed range.

Experimental investigation

Figure 8 compares the measured and predicted lateral displacement along the vertical axis of the housing at 4,800 rpm and shows a satisfactory experimental validation of the model. The improvement of the vibration level of the housing lies in the transformation of a conical shape (balancing A) into a cylindrical shape (balancing D). Therefore, the transmission of vibration to the pipes, which are connected at the top of the housing, is reduced. This has been checked throughout the 1,200-7,200 rpm speed range.

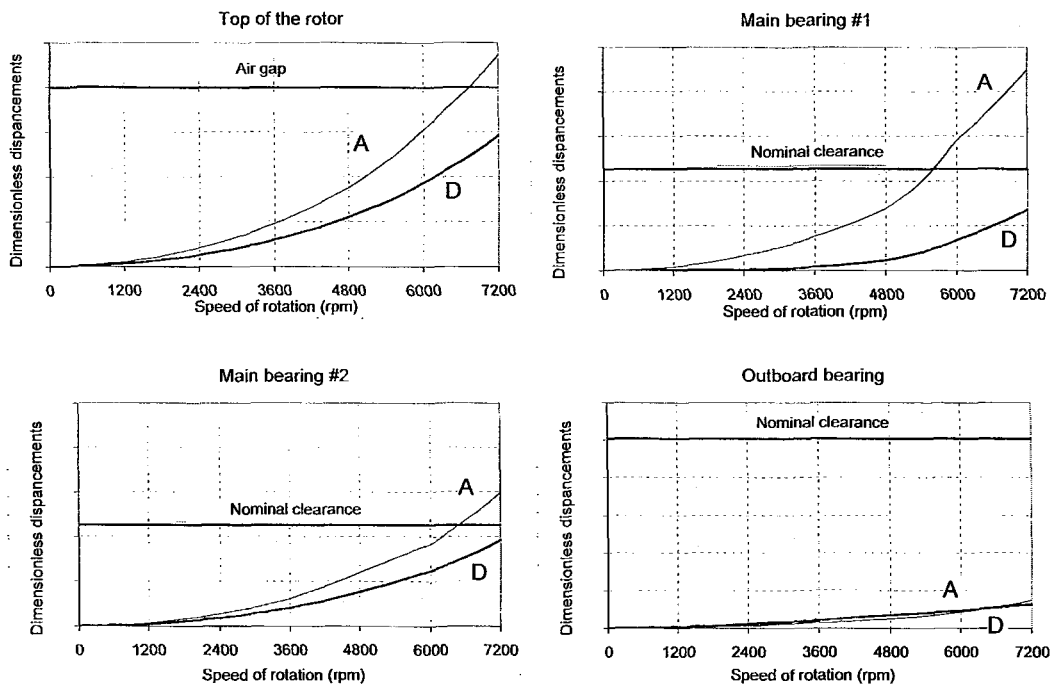


Figure 6: Displacements of the rotor - crankshaft – A and D balancings

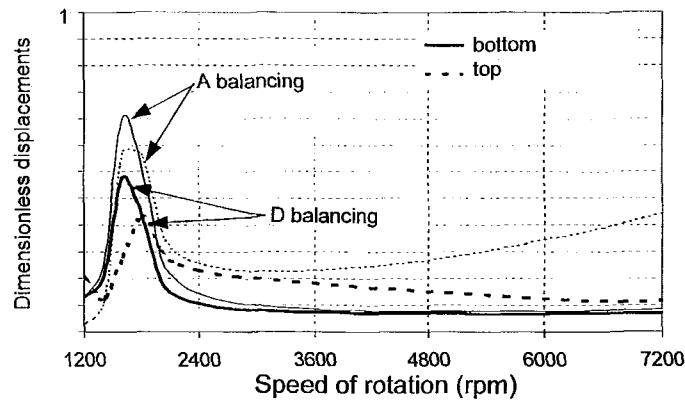


Figure 7: Displacements of the housing – A and D balancings

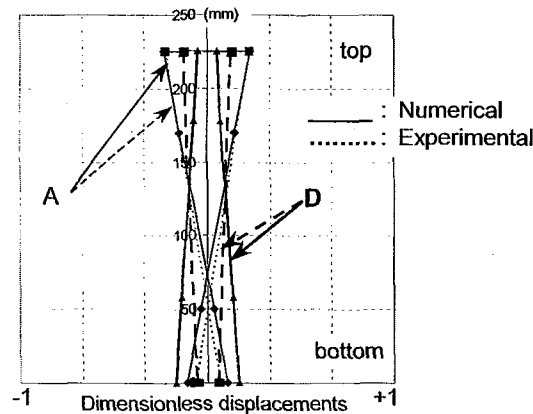


Figure 8: Dimensionless lateral displacement along the housing – A and D balancings

CONCLUSION

The balancing of a rotary compressor capable of variable speed of rotation was investigated numerically and validated experimentally. The prediction used an FE model based on the rotor-dynamics theory and the influence coefficient balancing method. In order to improve the current balancing, which is inadequate from 5,000 rpm upwards, several balancings with different criteria have been carried out.

Balancing B, which focuses only on the vibration level of the rotating part, reduces it substantially, but increases the vibration level of the housing.

Balancing C, which focuses only on the vibration level of the non-rotating part, is in fact equivalent to the current balancing A and therefore is not satisfactory.

Balancing D focuses on both the vibration levels of the rotating and non-rotating parts. It permits avoiding the vibration problem in the 1,200-7,200 rpm speed range, i.e. in particular the rotor-to-stator and crankshaft-to-bearing rubs. This balancing was chosen since because it can be easily implemented industrially.

ACKNOWLEDGEMENTS

The authors are indebted to Tecumseh Europe Co. for its support and permission to publish these works.

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